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Cylinder Pressure Characteristics of Turbocharged and Naturally Aspirated Diesel Engines

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Abstract

The article analyses the ways of air charge formation in the diesel engine as well as the principles of cylinder pressure change during the combustion process. As test objects, diesel engines of analogous constructional parameters but of different air charge have been chosen. These are a two cylinder, air-cooling and naturally aspirated diesel engine D120 and a three cylinder, liquid cooling and turbocharged diesel engine Valmet 320DS. The article presents the cylinder pressure characteristics of the named engines on different working regimes, and compares the principles of pressure change in the different positions of crank angle degrees. The study results are used in further improvement of the simulation model of the combustion process.

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Keywords: compression ignition engines; cylinder pressure characteristics; phases of the combustion process; engine test results

1. Introduction

Work capacity of an internal combustion engine depends on the perfection of the production technology. This is guaranteed by engine-producing company's implemented quality system [1]. At the same time, durability of the engine depends on the wearing rate of engine's subsystem. The length of the durability period is determined by preciseness of the production process of the subsystem and the use of simulation models in this process [2, 3]. In this article the cylinder pressure process in a diesel engine has been studied with the aim to improve the former

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simulation model of the combustion process [4]. A working calculation model shall simulate the process of combustion, depending on the crank angle degree. In controlling this process, it is important to know the classifications of the phases occurring in the combustion process and the principles of their change. This topic has been thoroughly analysed by Kegl et al. [5].

Improvement of internal combustion engines' combustion process is currently topical. This is caused by requirements of vehicle use efficiency and environmental safety restrictions. Vehicle use culture in its entirety is important here. The combustion process can be changed, incl. damaged by the wrong use of the vehicle's engine and by changing repair technology. In this article the sample characteristics of cylinder pressure changes and phases of combustion process of the turbocharged and naturally aspirated diesel engines are given. In order to measure the combustion process parameters in the cylinder, a specific measuring technology is needed. Such a technology is present at the test laboratory of Estonian University of Life Sciences [4]. In what follows, combustion process phases and characteristics are discussed, which form a basis for developing engines' simulation models.

Nomenclature

c_1	respective empirical constant on naturally aspirated engines
c_2	respective empirical constant on turbocharged engines
m_a	air charge mass in indicator diagram point *a*, where inlet valves close
$m_{a.1}$	respective air masses on naturally aspirated engines
$m_{a.2}$	respective air masses on turbocharged engines
n	polytropic exponent
n_e	crankshaft rotational speed
p_a	air charge pressure in point *a*
p_{env}	environment pressure
p_k	air charge pressure after turbocharged
T_a	air charge temperature in point *a*
T_{env}	environment temperature
T_k	air temperature emitted from the turbocharger
V_a	air charge mass in indicator diagram point *a* (inlet valves closes)
α	length of the studied phase
β_t	actual molecular change factor
ρ_{env}	environment density
ΔT_a	preheating of the intake air from the details of a warm engine
ΔT_k	precooling air charge temperature change

2. Materials and methods

2.1. Description of the combustion process

The occurrence of the combustion process and its characteristics are determined by the following constructive factors: a) the construction of the engine's combustion chamber; b) the way of directing the air charge; c) the way of injecting fuel and its quantity. In addition to the above mentioned, the combustion process is also influenced by the engine's working regimes and the physico-chemical composition and quality of the used diesel fuel. Diesel fuel contains different hydrocarbons of acyclic and cyclic chains. Most important of these, which influence directly the process of combustion, are alkanes, alkens, alkadienes, polycyclic aromatic hydrocarbons and cycloalkanes. The quality of diesel fuel is also influenced by the amount of accessories added to the fuel. Accessories are used in the diesel fuel which improve its cetane index, combustion stability, resistance to corrosion and lubrication capacity, enhance freezing point and resistance to frothing as well as other modifiers.

The cetane number characterizes the spontaneous combustion of the fuel and its quality combustion. The higher it is, the higher the diesel fuel ignites and the better are the drive properties of the engine. The high cetane number of the diesel fuel implies its high consistency of cycloalkanes and HC-class of alkanes. Also, diesel fuel of too high cetane number is not recommended. Its early flame centres initiate in case of direct injection blend formation the non-complete combustion of non-evaporated and prepared fuel, which entails decrease in the engine economy and a number of exhaust gases. Evaluating diesel fuel quality makes use of calculated cetane index, the test technology for determining the cetane number is lacking. The values of cetane number and calculated cetane index overlap if spontaneous combustion improving additives have not been added to the diesel fuel. The methodology of calculated cetane index has been presented in standard EN-ISO 4264:2007 [6].

2.2. Classification of the combustion process phases

The process of combustion is possible to be presented as phases. The time of combustion phases can be directed by diesel fuel composition and engine's constructive and exploitation factors. Different references provide three to five phases in the combustion process of diesel fuel [7, 8, 9].

Increase in cylinder pressure and the time of combustion phases with naturally aspirated and turbocharged engines depends on the air charge mass directed to the engine. This can be determined by equation which takes into account the engine's constructive parameters, the content and density of the input air charge, and which are influenced by pressure and temperature parameter values:

$$\boldsymbol{m_a = V_a \cdot \rho_{env} \cdot (p_a/p_{env}) \cdot (T_{env}/T_a)} \tag{1}$$

In case of diesel engines with a turbocharged, the environment temperature T_{env} is to be replaced with the air charge temperature emitted from the charger:

$$T_k = T_{env}(p_k/p_{env})^{(n-1)/n} \tag{2}$$

In indicator diagram's point *a* of the turbocharged diesel engine, the air charge temperature is

$$T_a = T_{env} + \Delta T_a \tag{3}$$

In case of a diesel engine with a turbo compressor and intake air charge cooling system, the following relation is valid between the temperatures

$$T_a = T_k + \Delta T_a - \Delta T_k \tag{4}$$

By inserting relations (2, 3 and 4) into equation (1), we get the formula for determining the intake air mass of turbocharger and naturally aspirated four-stroke engines. Considering the fact that air charge pressure in point *a,* on naturally aspirated four-stroke engines is $p_a = c_1 \cdot p_{env}$ and on turbocharged engines

$p_a = c_2 \cdot p_k$ we get the air mass on naturally aspirated engines

$$m_{a1} = c_1 \cdot V_a \cdot \rho_{env} \cdot (1 + T_{env}/\Delta T_a), \tag{5}$$

and the air mass on turbocharged engines

$$m_{a2} = c_2 \cdot V_a \cdot \rho_{env} \cdot (p_k/p_{env}) \cdot T_k/\left[T_a(p_k/p_{env})^{n-1/n}\right]. \tag{6}$$

The measurement data received with the device AVL Indimodul 621 indicate that in case of naturally aspirated direct injection engines, where all fuel is injected to the cylinder at once, combustion process phases are clearly distinguishable.

I – air-fuel mixture delay phase characterizes the spontaneous combustion delay of the air-fuel mixture. The composition of the air-fuel mixture formed in the cylinder has several components. The main component that influences the delay phase is diesel fuel and its auto-ignition temperature. The fuel does not ignite momentarily when entering the combustion chamber, but a certain ignition delay occurs. In this phase, the cylinder pressure increases, the ignition centres are prepared and the air-fuel mixture auto-ignites. The smaller the ignition delay, the smoother the combustion process and dynamic engine work. I−phase lasts from the moment of fuel injection to the combustion chamber (α_{inj}) until the beginning of combustion, i.e. until the discrepancy point of the combustion process pressure line (p_z) pressure process pressure line (p_c) (drawing 1). During this phase, diesel fuel injection, evaporation, blending with gas charge and the oxidation processes of the pre-flame occur. The time of the I−phase depends on the fuel's physico-chemical composition and blend forming method, environmental conditions and loading pressure, pressure level, the temperature of the working mixture as well as engine's load and speed regime. The time period of the delay phase is shorter than the time of the fuel injection process; based on test data, it lasts 0.8-2 ms on engine D120. The length of the delay phase is calculated with the equation:

$$\tau_l = a/6n_e \,. \tag{7}$$

In case of a large delay phase, more fuel arrives in the combustion chamber. In that case, the pressure of the combustion process and the rigidity of engine work increase rapidly. Therefore, the delay phase has to be as short as possible. The pressure increase limits that form in the delay phase and guarantee the normal functioning of the diesel engine is $d_p/d_a = 0.1 \ldots 0.2$ MPa deg^{-1}.

II – thermal balance phase. In a certain period of the pressure process, oxygen molecules decompose into atoms and move between hydrocarbon atoms, forming OH-group molecules, which increase the oxidation process. The more active components of them provoke the formation of nuclear centres in the reaction, and the combustion process begins. At the moment when the nuclear centres and spot fires are formed, a large part of the heat collects onto cylinder walls for evaporating the injected fuel. At this moment, the situation of critical temperature is formed in the combustion process, which causes thermal balance. The amounts of the heat formed in the combustion process and transferred to the combustion chamber are equalized. Characteristically, this process can be seen on p–α diagram, in the end of the delay phase (Fig. 1). The decrease of the function line characterizing the cylinder pressure change in this phase is related to heat use for evaporating the injected fuel spray. Thus, in II-phase, there exists an oxidation process the duration of which depends on the physico-chemical properties of the diesel fuel and the engine's mixture formation method. The thermal balance phase begins from the discrepancy point of the cylinder pressure lines and lasts until the moment of fast rise in pressure. Depending on the engine's working regime, II−phase lasts 0.2-0.5 ms.

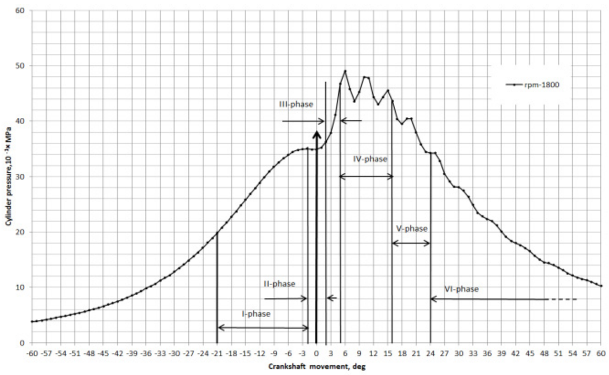

Fig. 1. The exemplary characteristic of naturally aspirated diesel engine D120 cylinder pressure on nominal speed.

III – quick pressure rise phase is characterized by quick rise in pressure in the engine's cylinder. There is an explosive development of ignition centres, and the combustion process encompasses the amount of the entire injected fuel. The quick pressure rise phase begins from the end of II‒phase and lasts until the moment where the maximum range of pressure rise stabilizes (Fig.1). The quality of this phase is described by the rake angle of the process's pressure rise line. During this phase, the one-injection injector of mechanical direct injection supply equipment is maximally open. During this phase, the main amount of the determined cyclical fuel feed is transferred to the engine. The length of III‒phase depends on the length of I‒ and II‒phase, fuel feed characteristics, the quality of fuel injection, the intensity of air charge movement as well as the engine's speed and load regime. During III‒phase, the piston is located in the top dead centre (TDC). A massive heat release begins, apparent combustion spreads, the average cyclic temperature and pressure increase. During the second and the third phase, the quality and the quantity of combustion products and change. This depends on the relation between a fresh working mixture and burnt gases.

Let us have in the indicator diagram point a the amount of fresh air charge in kilo moles M_a, while in point z the kilo mole amount of the burned gases M_z, and thus, during combustion, the working mixture's kilo mole amount change ΔM occurs. Taking into account the relation that

$$\Delta M = M_z - M_a$$

we can find the working mixture's chemical molecular change factor:

$$\beta_k = M_z/M_a = (M_a + \Delta M)/M_a = 1 + \Delta M/M_a, \tag{8}$$

at the same time, it is known that during combustion process, tail gases of the previous process with the kilo mole amount of M_r are added to the working mixture. We can present the previous relation in the following way

$$\beta_t = (M_z + M_r)/M_a, \tag{9}$$

As the relation M_r/M_a expresses the tail gas factor γ_r and M_z/M_a expresses chemical molecular change factor, we can present the equation (10) in the following way:

$$\beta_t = (\beta_o + \gamma_r), \tag{10}$$

Factors β_k and β_t are parameters that influence the relations between actual pressure and the cylinder pressure received during the simulation model [2].

IV – main combustion phase. This is the major combustion period of the working mixture. IV‒ phase begins from the end of III‒phase and lasts during the entire stabilization area of the combustion pressure. The end of the phase corresponds to achieving the maximal combustion temperature or until the formation of a quick decrease in the combustion pressure. This phase is characterized by the spreading of the flame, formed from the combustion zones, to the entire space of the combustion chamber, achieving thereby quick volumetric combustion. The increasing spreading of the flame and the amount of the combustion final products increases the combustion temperature to the maximum. The main combustion phase lasts depending on the engine's load regime until the crank shaft angle 10…15 degree after TDC. The increasing volume of the combustion chamber and the increasing combustion pressure achieve an approximate balance; therefore, the engine's dynamic load does not practically increase and reaches limits where $dp/d\alpha = 0.03\text{-}0.04$ MPa deg^{-1}. In the main combustion phase, the most efficient work is done in the cycle.

V – quick pressure decrease phase begins from the final point of the combustion pressure (IV‒phase end) and lasts until the pressure decrease reaches the cylinder compression value p_c, where the combustion process practically ends. The quick pressure decrease phase is caused by the end of combustion of the working mixture and a significant increase in cylinder combustion space due to the moving of the crankshaft. Here, a relation is at work, where the elementary change of the linear measurement of the piston moving triples the increase of the elementary change of the cylinder volume. The length of the quick pressure decrease, depending on the engine work mode, remains in the limits of the crank shaft angle of 5-11 degrees. The speed of the post-combustion process of V− and VI–phase is influenced by the size of the fresh air charge mass, directly injected to the cylinder, and the turbulent mixing speed of non-complete combustion products. The extension of this phase raises exhaust gas temperature and their smokiness, as well as decreases the engine's efficiency. The pressure change speed is in the limits of $dp/d\alpha$ = 0.07-0.14 MPa deg^{-1}.

VI – slow pressure decrease phase forms a significant part of the expansion process. The phase begins from the final moment of the quick pressure decrease and lasts until the end of the expansion process (on engine D120 up to 140 degrees before the bottom dead centre - BDC). Pressure in the cylinder decreases quickly. The pressure and length of the combustion products of this phase determine which exhaust gases and with which parameters exit from the engine and what is the efficiency the entailed energy use. Optimize of the pressure change of the diesel fuel combustion process influences the rise of pressure and changes the smooth work of the engine. The main influences on the combustion process phases are combustion time, temperature and the physico-chemical composition of the fuel. Combustion time and temperature shorten, and the fuel quality lengthens the combustion phases. Extension of this phase raises the temperature and smokiness of exhaust gases, and decreases engine efficiency. The time of this phase has to be minimized. The average time of VI−phase is 30º−40º and pressure change speed, dp/dα = 0.06-0.08 MPa deg^{-1}. Regardless of the engine working regime, in the presented phase, cylinder pressure parameters are stabile and change according t the same principle.

As studies indicate [2], in quick and slow pressure decrease phases, the functional relation of the cylinder pressure change differs significantly from the function received with a simulation model. The cylinder pressure received with the latter is higher than the actual cylinder pressure. This has been partly caused by the imprecision of the working gases actual molecular change factor as well as heat use imprecision, but also by the choice of the universal gas state constant value.

2.3. The description and technical data of the test object and measuring instruments

The test objects were diesel engines Valmet 320DS and D120 (Table 1). Both engines were specially adjusted for measuring cylinder pressure. On diesel engine Valmet 320DS, pressure gauge AVL GH 13P was placed on the cylinder head in the manufacturer's factory; on the engine D120, pressure gauge Kistler 701A was used (Table 2).

Table 1. Diesel engines technical specifications.

Gauges and technical

Table 2. adapters

Parameter	Valmet 320DS	D120
In-line	turbocharged	natural aspirated
Number of cylinders	3	2
Cylinder diameter	108 mm	105 mm
Piston stroke	120 mm	120 mm
Volume	3.3 liter	2.08 liter
Combustion chamber	non-divided	non-divided
Fuel injection pump	in-line	in-line
Pump fuel delivery	96 mm^3 $stroke^{-1}$	59 ± 2 mm^3 $stroke^{-1}$
Fuel injection angle	19 deg BTDC	21 deg BTDC
Power	53.5 kW	18.4 kW
Maximum torque	285 Nm	99.5 Nm
Pressure ratio	16.5	16.5
Nominal rotational speed	2,400 rpm	1,800 rpm
Maximum torque achieved	1,450 rpm	1,260 – 1,400 rpm
Maximum rotational speed	2,625 rpm	1,950 rpm
Idling speed	750 rpm	800 – 1,050 rpm
Specific fuel consumption	269 g kWh^{-1}	245 g kWh^{-1}
Fuel consumption nominal	14.4 kg h^{-1}	6.37 kg h^{-1}
Cooling system	water-cooling	air-cooled
Compression	2.4 PMa	3.0 MPa
Inlet/exhaust valve seat bore diameter	48/41 mm	44/38 mm
Inlet/exhaust valve max lift	10.9 / 12.1 mm	11.6 mm
Inlet/ exhaust valve angle of slope	35,2 /45,2 ^{0}	45 ^{0}
Inlet valve opens/closes	$0^0 \pm 2^0$ BTDC/$16^0 \pm 2^0$ ABDC	16^0 BTDC/40^0 ABDC
Exhaust valve opens/closes	$39^0 \pm 2^0$ BBDC/$1^0 \pm 2^0$ATDC	40^0 BBDC/16^0ATDC

specifications.

Gauge	Main parameters	Adapter
AVL GH 13P	measure range 0… 25.0 MPa	no adapters
	operating temperature range − 40…400 ^{0}C	diameter of hole - M5×0,5
	pieso-input cable C131-1	-
Kistler 701A	measure range 0… 25.0 MPa	outside diameter - tube M 12×1
	operating temperature range −150…200 ^{0}C	length of thread -min 12 mm
	connector, teflon insulator 10-32 UNF	-

2.4. Description of test method

Test engines were installed on a dynamometer Schenck 3-LI 250, equipped with a frequency converter. In addition to the measurement instruments indicated in Table 2, the engines were equipped with different temperature, pressure and rotation frequency gauges as well as with air and fuel consumption measuring devices (Table 3). Summer and winter diesel fuel of five different retail companies was used on the test engines, altogether eight variants. On the purposes of the study, standard load and regulation characteristics were conducted with the test engines. Measurements of cylinder pressure and the following analysis of the measurement results were done with the device AVL Indimodul 621.

Table 3. Fuel Mass Flow Meter AVL 7351 CME technical specifications.

Recommended measuring range	0 ... 125 kg/h, 0 ... 165 l/h (at 0.75 g/cm³)
Systematic measurement uncertainty	U_s = 0.12%
Density measurement uncertainty	≤ 0.0005 g/cm³
Ambient temperature	0 ... 50 °C
Fuel supply pressure	0.1 ... 0.8 bar
Fuel supply flow	max. consumption + 20 kg/h
Fuel supply temperature	- 10 ... + 40 °C
Outlet pressure	adjustable from 0.05 ... 0.5 bar
Dimensions	770 x 670 x 345 mm (W x H x D)
Weight (dry)	62 kg

3. Results and discussion

The study results of actual combustion processes

Diesel engine’s D120 cylinder pressure changes in pressure and combustion process are the following:

1. the cylinders’ average pressure process pressure p_c by measuring with a compression tester, on the rotational frequency 400 min^{-1} it is p_c = 3.6 MPa;

2. on the rotational frequency of 1,800 min^{-1} the mean value of the maximum pressure of the cylinders’ pressure process is p_c = 3.6 MPa;

3. the changes of cylinder Nr1 pressure in the combustion process during the measurement of the regulatory characteristic:

a) on speed modes of a smaller load (up to V_f = 65 mm^3 per cycle^{-1}) up to TDC, the cylinder pressure does not practically change and is equal to the pressure of the pressure process (p_c = 3.5-3.6 MPa);

b) on speed modes of average load (V_f= 65-75 mm^3 per cycle^{-1}), the combustion process pressure of the cylinder up to 3 degrees before the TDC does not basically change and is equal to the pressure of the pressure process;

c) on big modes loads (V_f = 75-80 mm^3/per cycle), the cylinder pressure decreases noticeably before the TDC and acquires values in the limits of p_c = 30-32 bar. The process of cylinder pressure change on engine D120 has been submitted on Figure 1 and the respective data in Table 5.

Grouping of the cylinder pressure received on the different working regimes of the diesel engine D120 (Fig. 2).

On engine D120, a group of load and regulatory characteristics were conducted, during which the cylinder pressure was respectively measured. During the regulatory characteristics presented in the article, measurements were conducted on 15 speed regimes. On this engine’s regulatory characteristics, speed regime groups distinctively form, in the course of which cylinder pressure changes acquire approximate vicinity. These groups are:

a) group 1, of high cylinder pressure value p_z = 5.5-5.7 MPa. Here belong regimes: n_e = 950 min^{-1}; n_e = 1,000 min^{-1}; n_e = 1,100 min^{-1}; n_e= 1,300 min^{-1}. The characteristic regime of the group is: n_e = 1,000 min^{-1};

b) group 2, of high cylinder pressure value p_z = 5.1-5.4 MPa. Here belong regimes: n_e = 1,200 min^{-1}; n_e = 1,400 min^{-1}; n_e = 1,500 min^{-1}. The characteristic regime of the group is: n_e = 1,500 min^{-1};

c) group 3, of average cylinder pressure value p_z = 4.4-4.8 MPa. Here belong regimes: n_e = 1,600 min^{-1}; n_e = 1,700 min^{-1}; n_e = 1,750 min^{-1}; n_e = 1,800 min^{-1}; n_e = 1,823 min^{-1}. The characteristic regime of the group is: n_e = 1,800 min^{-1};

d) group 4, of low cylinder pressure value p_z = 3.6-4.0 MPa. Here belong regimes: n_e = 1,841 min^{-1}; n_e = 1,858 min^{-1}; n_e = 1,870 min^{-1}. The characteristic regime of the group is: n_e = 1,858 min^{-1}.

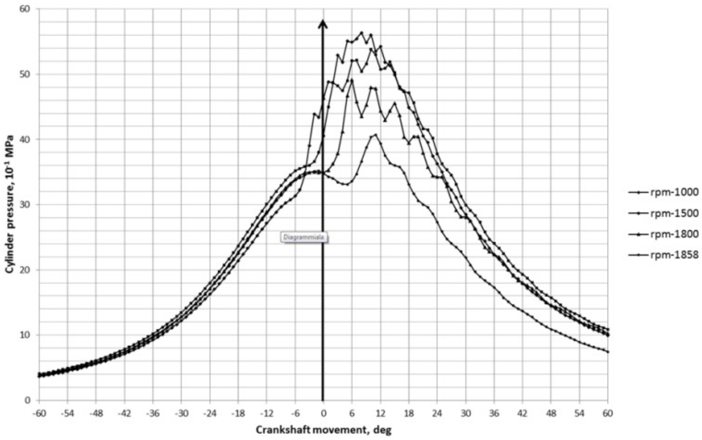

Fig. 2. Cylinder pressure characteristics of diesel engine D120 on the characteristic modes.

The phases of the diesel engine D120 combustion process are classified on the basis of the above presented theory (table 4).

Table 4. Pressure and phase parameters of the engine D120 on different speed regimes.

Mode group	Parameter	I-delay-phase, deg	II-thermal balance phase, deg	III-quick pressure rise phase, deg	IV- main combustion phase,deg	V- quick pressure decrease phase, deg	VI- slow pressure decrease phase, deg
Group 1: modes n_e= 900…1100 min^{-1}	Phase start	−21.0	−7.0	−5.0	+2.0	+16.0	+25.0
	Phase end	−7.0	−5.0	+2.0	+16.0	+25.0	+60.0
	Initial pressure	20.0	30.8	32.2	49.0	48.0	35.0
	Final pressure	30.8	32.2	49.0	48.0	35.0	10.0
	mute dp/dα	0,77	0,70	2.4	-	1.44	0.71
Group 2: mode n_e= 1200…1500 min^{-1}	Phase start	−21.0	−5.0	−3.0	+1.0	+16.0	+27.0
	Phase end	−5.0	−3.0	+1.0	+16.0	+27.0	+60.0
	Initial pressure	20.0	35.7	36.0	49.0	48.0	35.0
	Final pressure	35.7	36.0	49.0	48.0	35.0	10.0
	mute dp/dα	0.98	0,15	3.25	-	1.44	0.76
Group 3: mode n_e= 1600…1800 min^{-1}	Phase start	−21.0	−3.0	0.0	+5.0	+17.0	+24.0
	Phase end	−3.0	0.0	+5.0	+17.0	+24.0	+60.0
	Initial pressure	20.0	35.0	35.0	47.0	40.0	35.0
	Final pressure	35.0	35.0	47.0	40.0	35.0	10.0
	mute dp/dα	0,83	0,0	2.4	-	0.71	0.69
Group 4: mode n_e= 1800…1858 min^{-1}	Phase start	−21.0	−2.0	+5.0	+10.0	+12.0	+17.0
	Phase end	−2.0	+5.0	+10.0	+12.0	+17.0	+60.0
	Initial pressure	20.0	35.0	33.0	40.0	40.0	35.0
	Final pressure	35.0	33.0	40.0	40.0	35.0	8.0
	mute dp/dα	0.79	0.29	1.4	-	1.0	0.62

Diesel engine Valmet 320DS cylinder pressure changes in the pressure and combustion process are the following:

1) cylinder compressions were measured on the rotational frequency of the engine's starter; the average pressure of the three measuring results are $p_{c.max}$ = 24 bar;

2) on the rotational frequency n_e = 2,400 min^{-1} the mean value of the maximum pressure of the cylinders' pressure process was p_c = 36.0 bar;

3) during the research, four regulatory characteristics were conducted;

4) the regulatory characteristic presented in the article was conducted on 11 speed modes and the pressure values of cylinder nr 1 were measured; results have been indicated in Table 5 and Figure 3;

5) cylinder pressure characteristics on all rotational speeds have been shifted ca 11 degrees before the TDC;

6) at the increase of the engine's rotational frequency, i.e. in the limits of n_e = 1,100−2,400 min^{-1} cylinder pressure increases and they acquire the principle of a same change;

7) at the continuous rise of the engine's rotational frequency, i.e. on modes n_e = 2,480−2,610 min^{-1} cylinder pressure decreases on the same principle;

8) at the engine's rotational speed modes of n_e = 1,800−1,100 min^{-1}, i.e. toward the increase of the engine load, the maximum value of the cylinder pressure shifts toward the increase of the combustion chamber volume;

9) pressure change value (Table 5) handles the change between pressure and phase from the moment of fuel injection (19 crank angle degrees before the TDC) until the maximum value of the combustion pressure of the respective regime.

Table 5. Pressure and phase parameters of engine Valmet 320DS on different speed modes.

Mode, min^{-1}	Initial pressure, bar	Max combustion pressure, bar	Max combustion pressure, deg	Pressure change, bar deg^{-1}
n_e = 1100	40.0	77.0	+13.0	1.16
n_e = 1250	44.0	80.0	+12.0	1.16
n_e = 1450	46.0	78.2	+11.0	1.07
n_e = 1600	49.0	80.0	+12.0	1.00
n_e = 1800	53.0	84.0	0.0	1.63
n_e = 2000	56.0	89.0	-2.0	1.94
n_e = 2400	60.0	95.0	-2.0	2.06
n_e = 2480	54.0	86.0	-2.0	1.88
n_e = 2540	46.0	73.0	-2.0	1.59
n_e = 2600	36.0	58.0	-2.0	1.29
n_e = 2610	34.0	55.0	-2.0	1.24

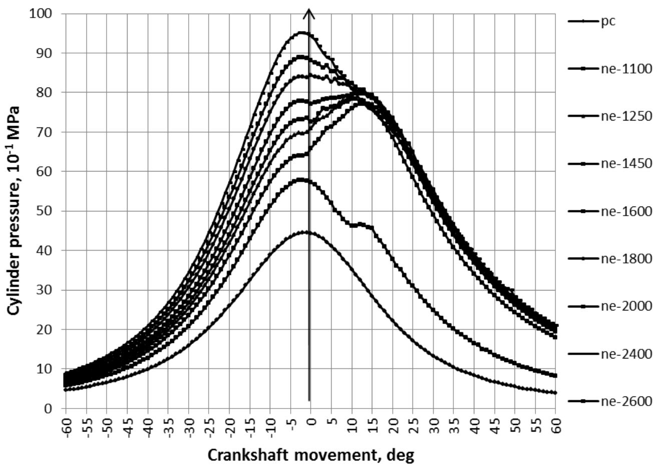

Fig. 3. Cylinder pressure characteristics of the diesel engine Valmet 320DS.

Conclusions

The aim of the study was to work out the characteristics of cylinder pressure changes for two diesel engines of different air charging systems, and to compare the phases of their combustion processes. The resultant characteristics will, henceforth, serve as the basis for designing and developing cylinder pressure simulation models of internal combustion engines.

Summary of research results as for naturally aspirated diesel engine D120

1. The combustion process phases of the diesel engine D120 are clearly distinguishable from each other, and there is the total of at least six phases.
2. The greater the load, the greater the cycle's fuel feed and the smaller the delay phase. Therefore, load does influence the time length of the I-phase and the pressure of the combustion process occurring during the phase.
3. When decreasing the load and increasing the rotational frequency, the length of the delay phase remains on all regimes approximately the same.
4. The length of the thermal balance phase depends on the engine's speed and load mode.
5. The length of the quick pressure rise phase is influenced by the load mode and the time allotted for the fuel combustion.
6. The length of the main combustion phase is influenced above all by the load and speed modes.
7. The length of the quick and slow pressure decrease phases is influenced by the load and speed modes

Summary of research results as for the turbocharged diesel engine Valmet 320DS

1. The cylinder pressure of Valmet 320DS is about two times higher than the cylinder pressure of D120.
2. The mass of the fresh air charge directed to the engine has not been co-ordinated with the turbo compressor capacity of the specific (nominal) mode of engine operation.
3. The maximum cylinder pressure of the nominal mode is before top dead centre.
4. In higher speed modes, the cylinder pressure decreases proportionally with the decrease in the load mode.
5. The construction of the turbocharger of the engine Valmet 320DS needs an improvement by using a pressure regulation device.

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